3746

PATENT APPLICATION
Atty. Docket No. 3419-990811

#6

IN THE UNITED STATES PATENT AND TRADEMARK OFFICE

Group Art Unit 3746

In re application of

J. MICHAEL TEETS ET AL. : **AN ELECTRICITY GENERATING SYSTEM HAVING AN ANNULAR COMBUSTOR**

Serial No. 09/319,478 :

Filed August 18, 1999 :

OIPE JC68 OCT 2 6 2000 PATENT & TRADEMARK OFFICE

Pittsburgh, Pennsylvania
October 23, 2000

RECEIVED NOV -2 2000 TC 3700 MAIL ROOM

CLAIM FOR PRIORITY UNDER 35 U.S.C. §119

Box Patent Application
Assistant Commissioner for Patents
Washington D.C. 20231

Sir:

Attached hereto is a certified copy of U.S. Provisional Application Serial No. 60/032,090, which corresponds to the above-identified United States application and which was filed in the United States Patent Office on December 3, 1996.

The priority benefits provided by Section 119 of the Patent Act of 1952 are claimed for this application.

Respectfully submitted,

WEBB ZIESENHEIM LOGSDON
ORKIN & HANSON, P.C.

By Richard L. Byrne

Richard L. Byrne, Reg. No. 28,498
Attorney for Applicants
700 Koppers Building
436 Seventh Avenue
Pittsburgh, PA 15219-1818
Telephone: 412/471-8815
Facsimile: 412/471-4094

PA 303512

THE UNITED STATES OF AMERICA

TO ALL TO WHOM THESE PRESENTS SHALL COME:

UNITED STATES DEPARTMENT OF COMMERCE

United States Patent and Trademark Office

September 26, 2000

THIS IS TO CERTIFY THAT ANNEXED HERETO IS A TRUE COPY FROM THE RECORDS OF THE UNITED STATES PATENT AND TRADEMARK OFFICE OF THOSE PAPERS OF THE BELOW IDENTIFIED PATENT APPLICATION THAT MET THE REQUIREMENTS TO BE GRANTED A FILING DATE UNDER 35 USC 111.

APPLICATION NUMBER: ***60/032,090***
FILING DATE: ***December 03, 1996***

PATENT AND TRADEMARK OFFICE
UNITED STATES OF AMERICA

By Authority of the
COMMISSIONER OF PATENTS AND TRADEMARKS

N. Williams

N. WILLIAMS
Certifying Officer

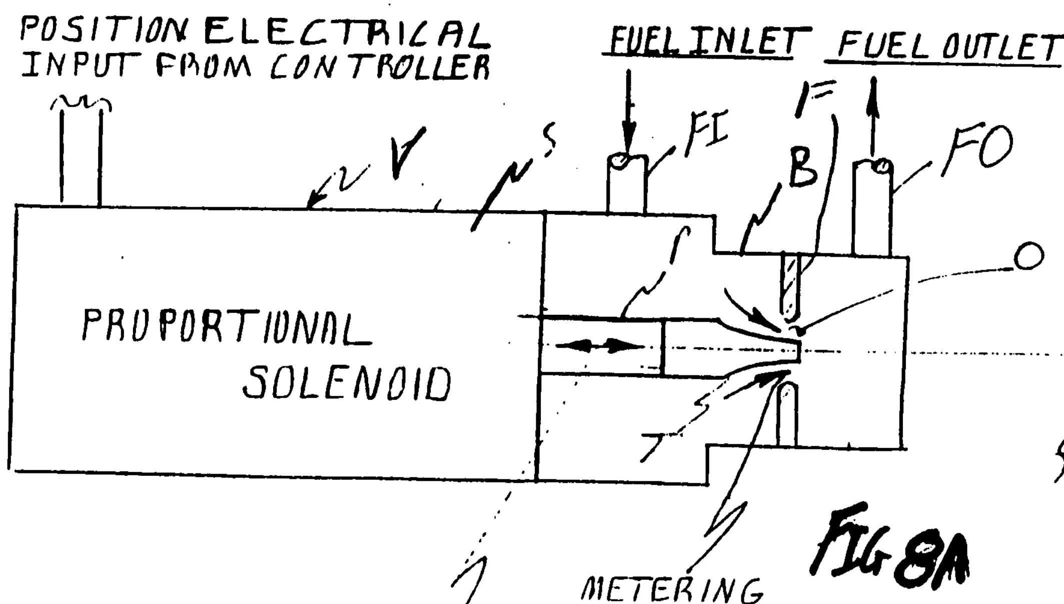


FIG 8A

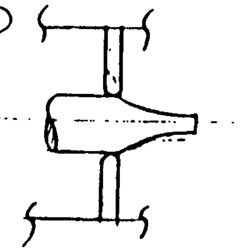


FIG 8B

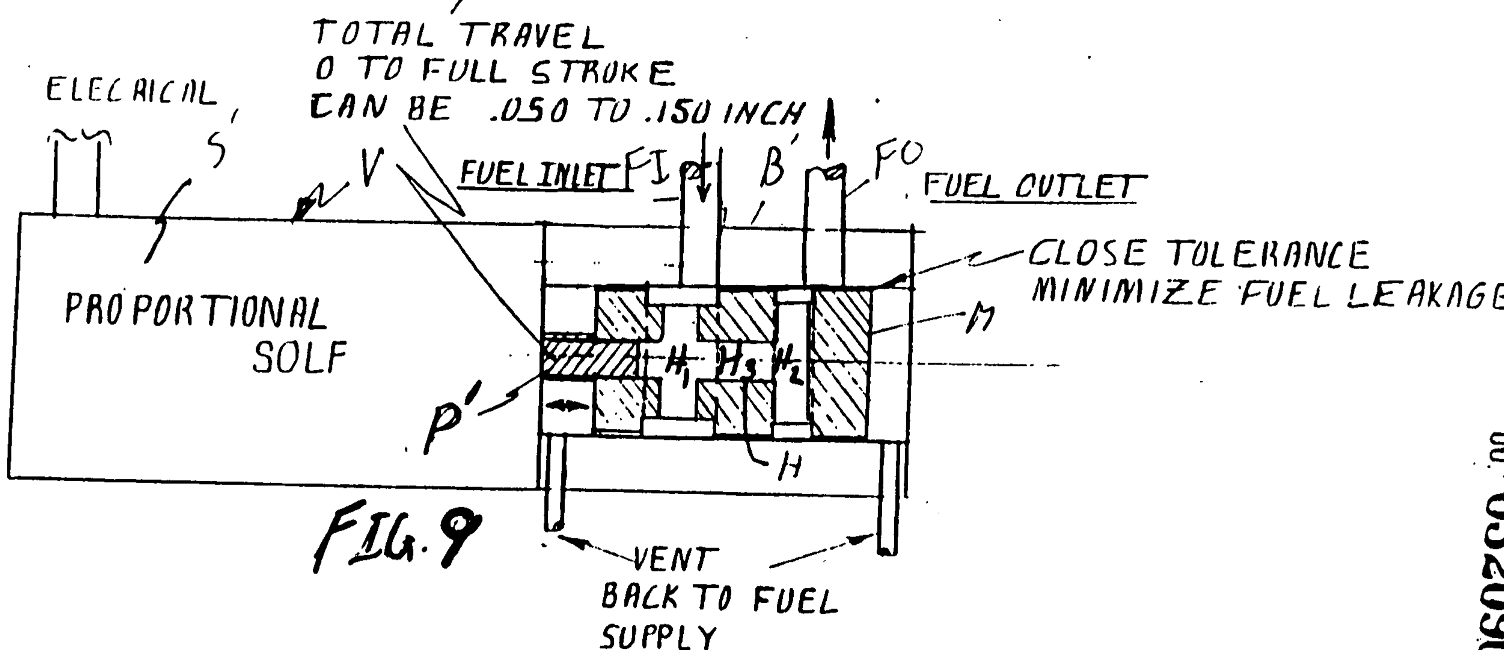


FIG. 9

60/032090

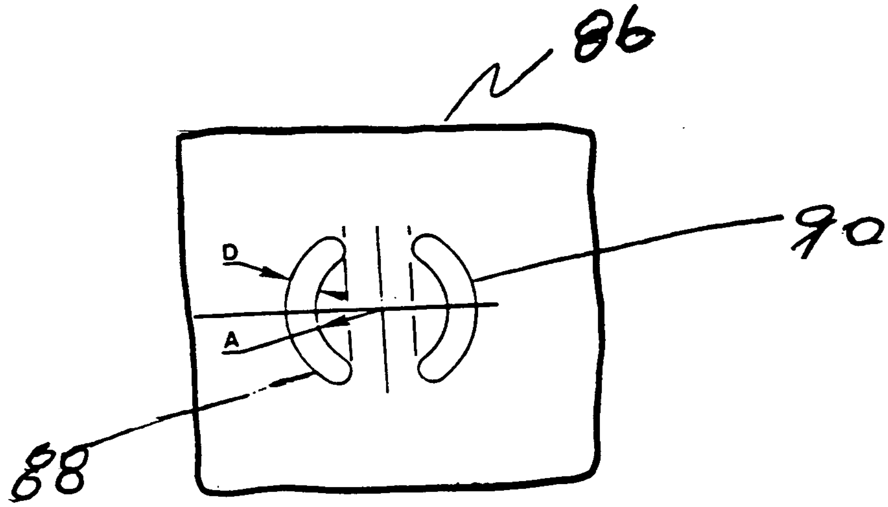
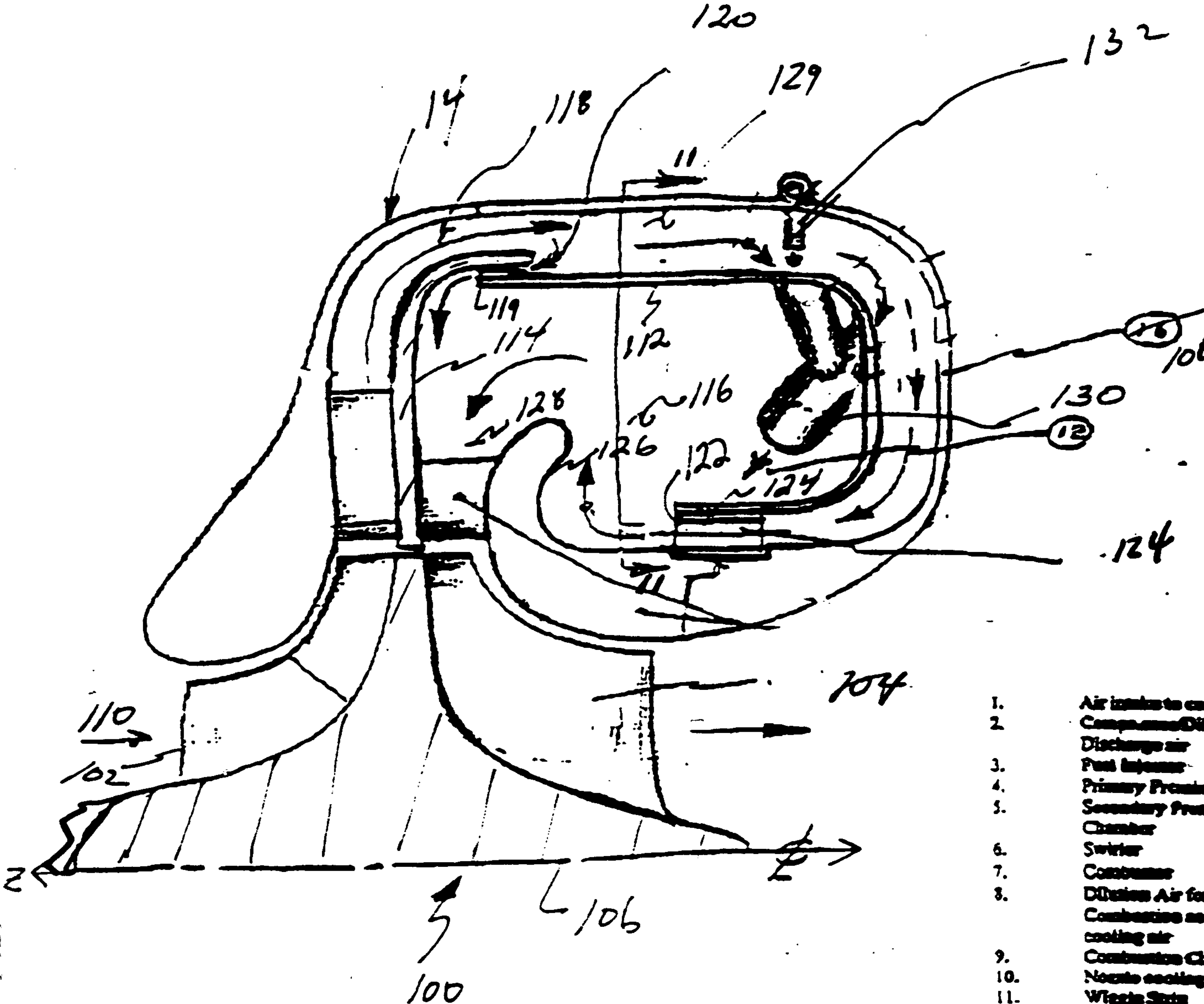


FIG 7

[redacted]

GAS TURBINE COMBUSTOR
STAGED PREMIX PREVAPORIZED LOW EMISSION

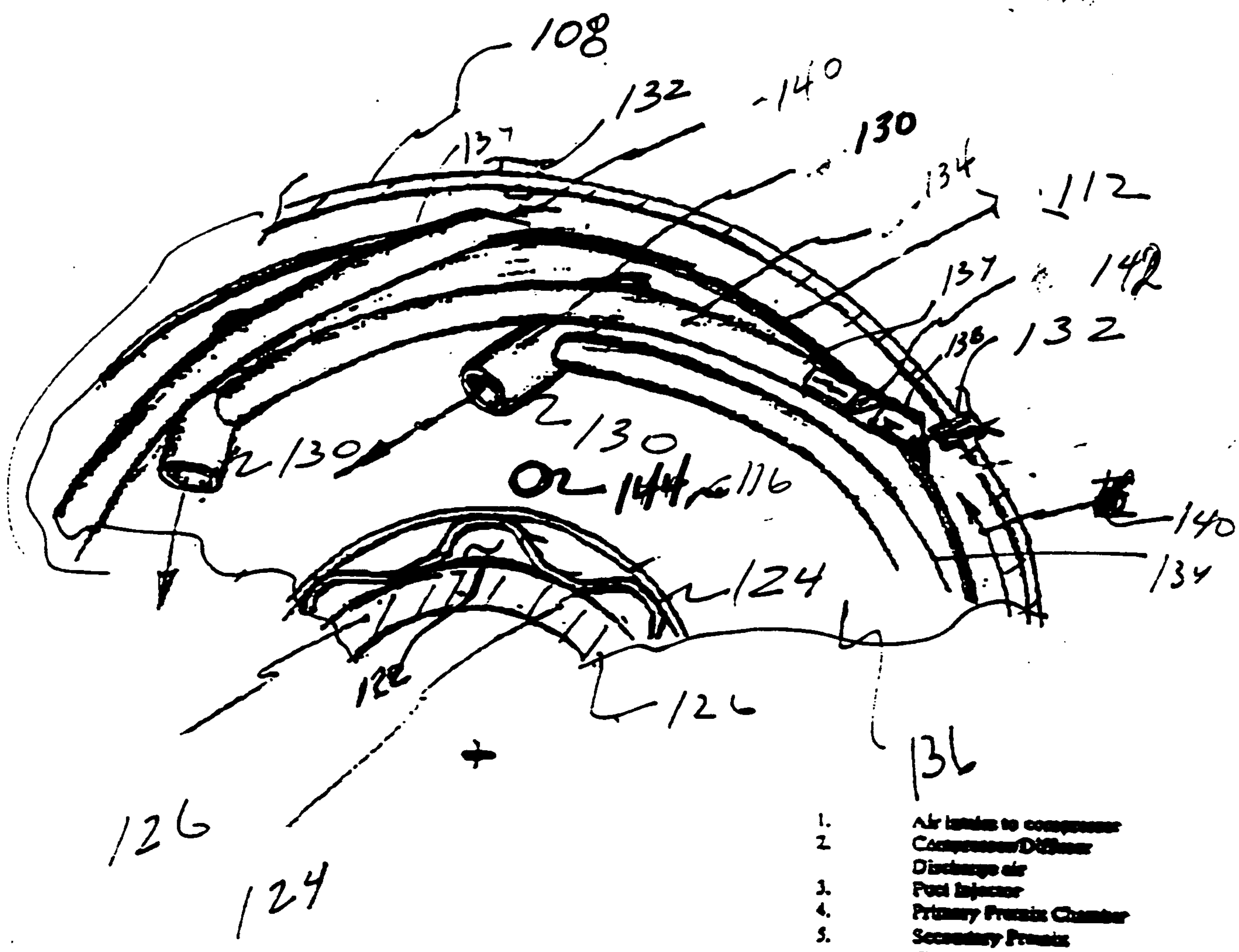
60/032090



1. Air intake to compressor
2. Compressor/Diffuser Discharge air
3. Fuel Injector
4. Primary Premix Chamber
5. Secondary Premix Chamber
6. Swirler
7. Combustor
8. Dilution Air for Combustion and Nozzle cooling air
9. Combustion Chamber
10. Nozzle cooling air
11. Wiggle Strip
12. Secondary Premix Discharge
13. Turbine Wheel
14. Turbine Exhaust
15. Turbine Nozzle
16. Combustor Housing
17. Primary Premix Air
18. Secondary Premix Air

FIG. 10

[redacted]



1. Air intake to compressor
2. Compressor/Diffuser Discharge air
3. Fuel Injector
4. Primary Premix Chamber
5. Secondary Premix Chamber
6. Swirler
7. Combustor
8. Dilution Air for Combustion and Nozzle cooling air
9. Combustion Chamber
10. Nozzle cooling air
11. Wiggle Strip
12. Secondary Premix Discharge
13. Turbine Wheel
14. Turbine Exhaust
15. Turbine Nozzle
16. Combustor Housing
17. Primary Premix Air
18. Secondary Premix Air

FIG. 4

~~FIGURE~~

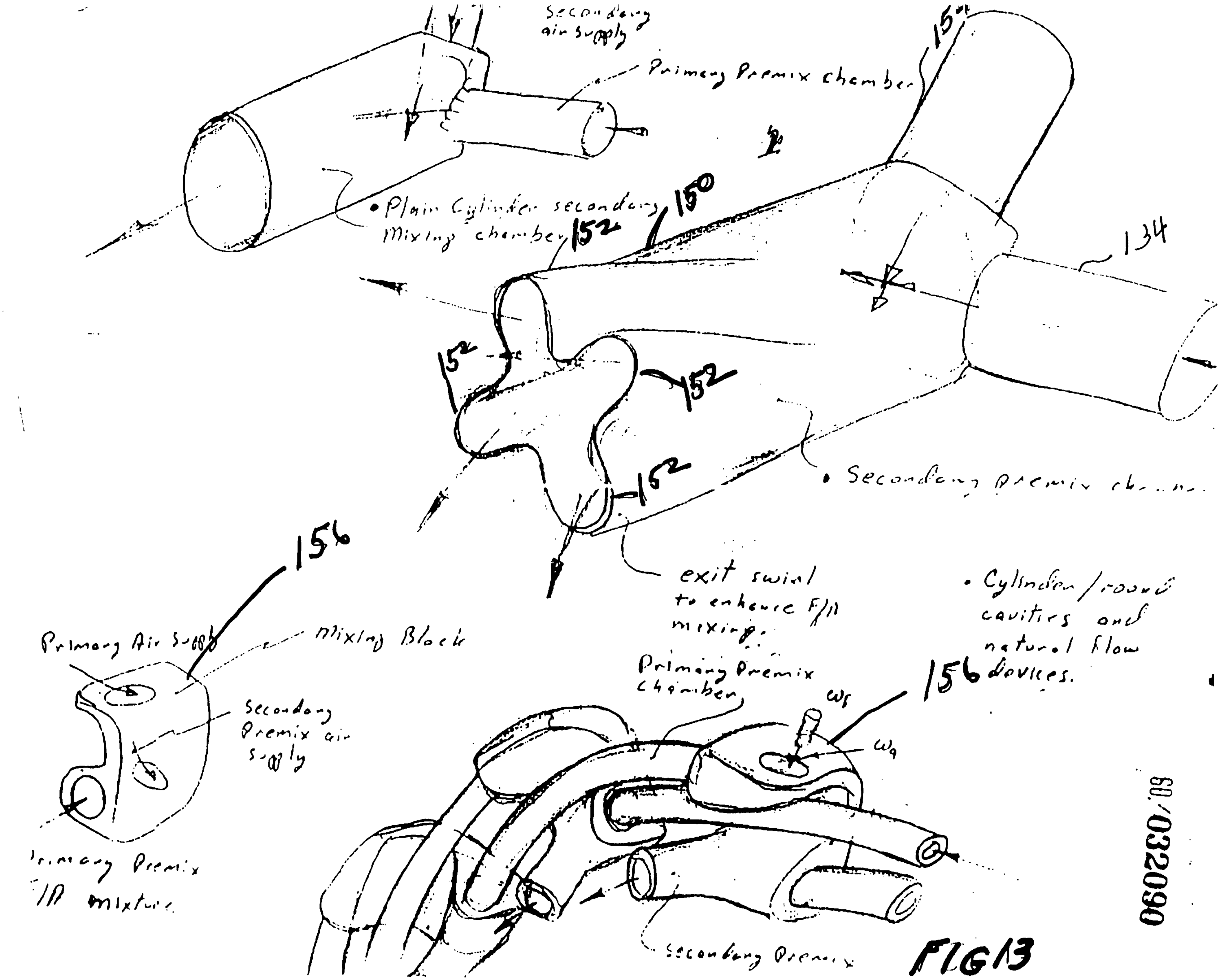


FIG 13

60/032090

60/032090

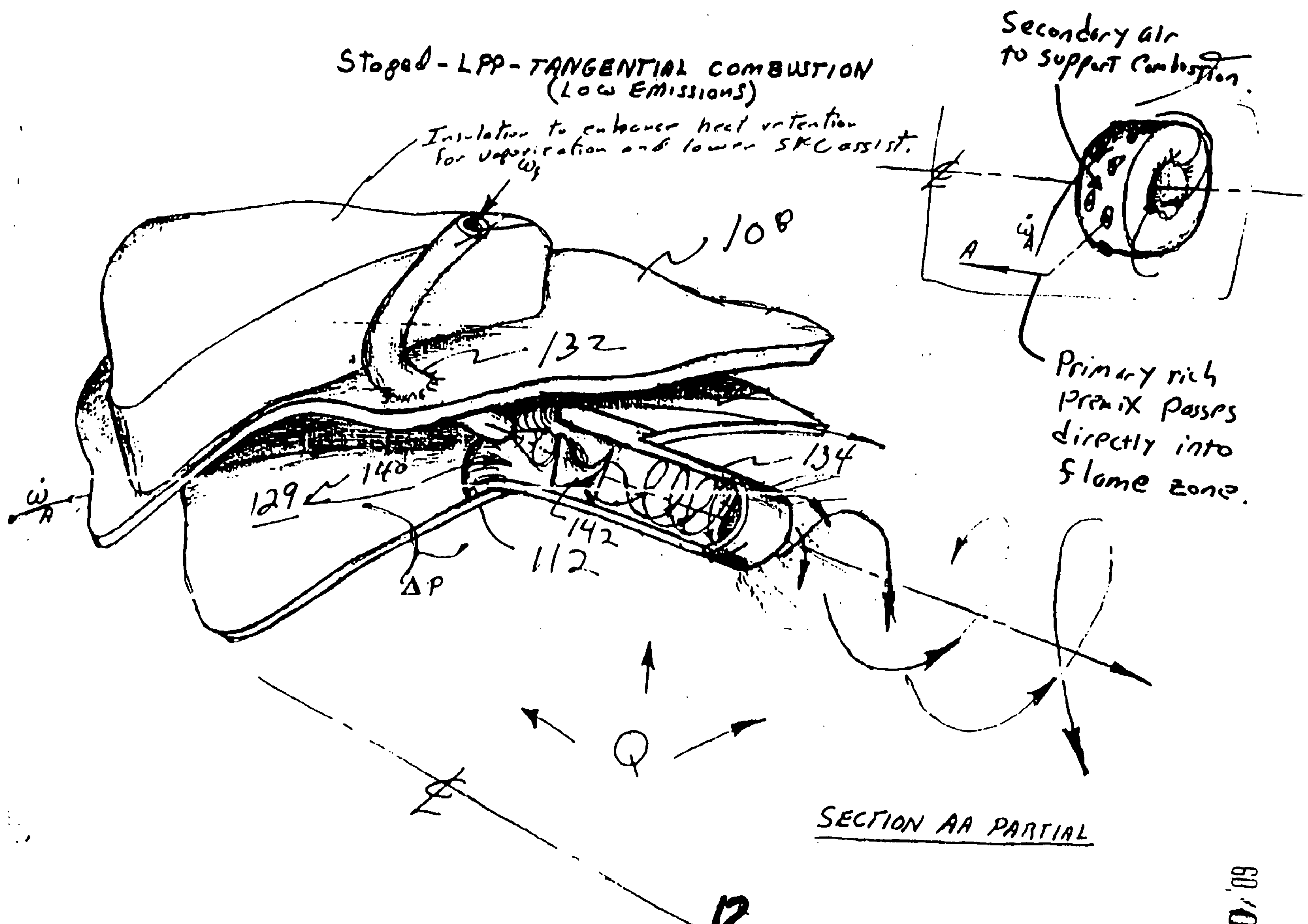
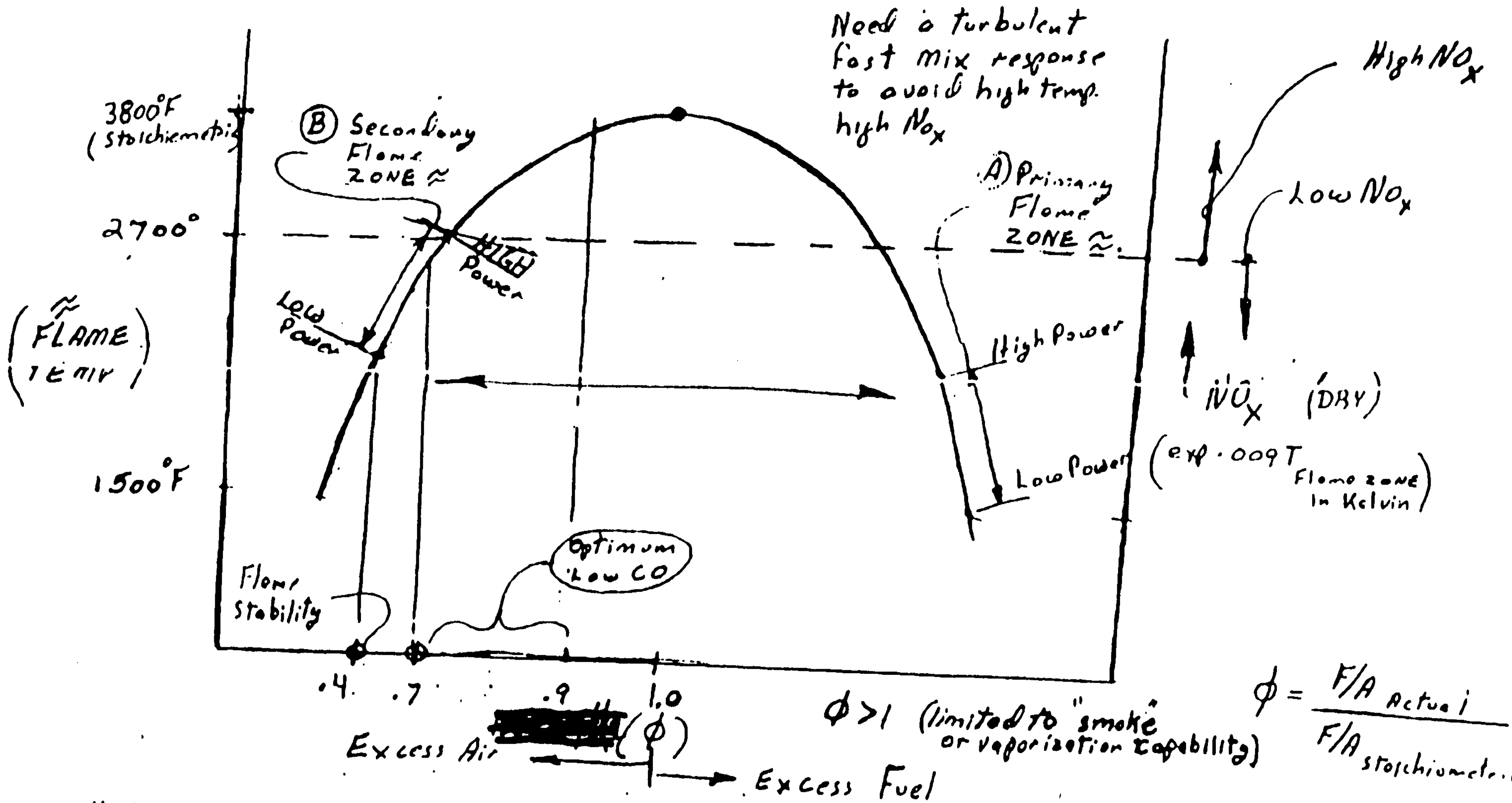


FIG. 12

FIG. [redacted] 04

Staged-LPP-TANGENTIAL COMBUSTION
(Estimated)



- High end ≈2100°F Flame temp. per figure 5 and 1800°F for the low end as a comprimise for low CO. Lefebvre infers the temp. range should be higher. Temp range depends on combustor configuration, fuel type and fuel preparation prior to flame.
- LOW UHC via efficient Combustor, vaporization and homogeneous F/A mixture in primary zone

Pg 10

60 032090

10 60,'032090

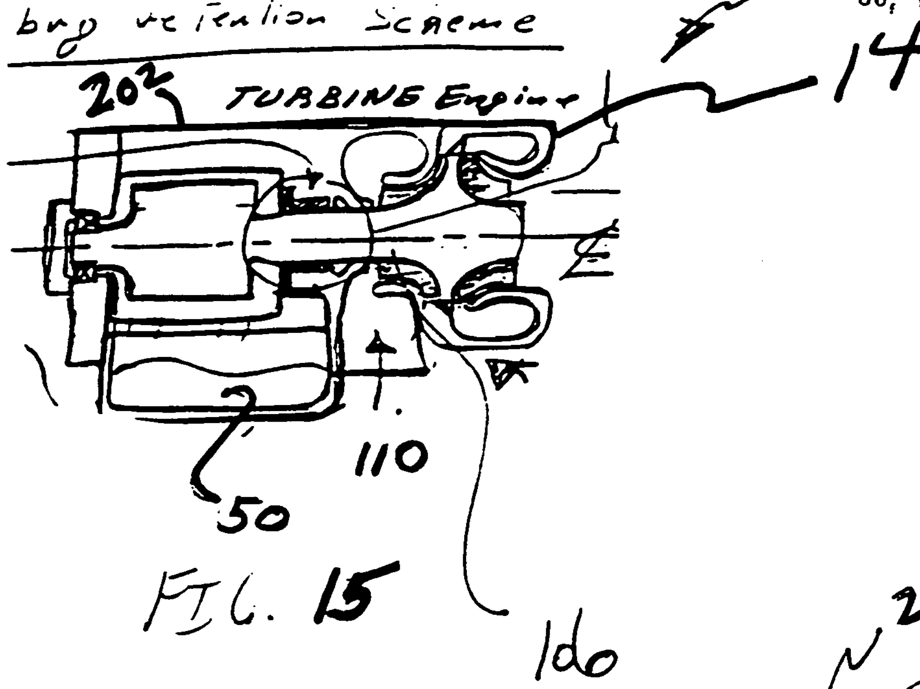


FIG. 15

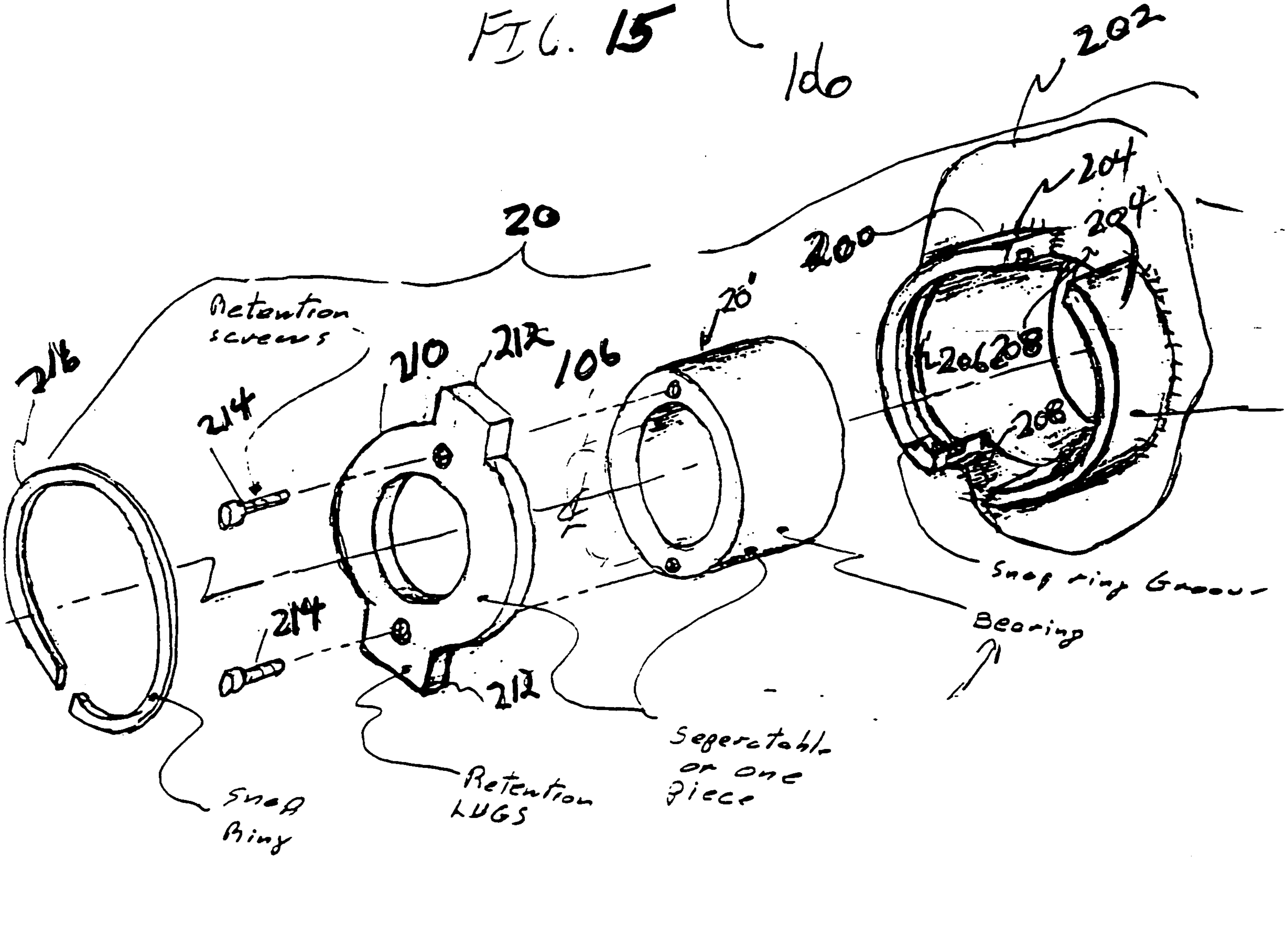
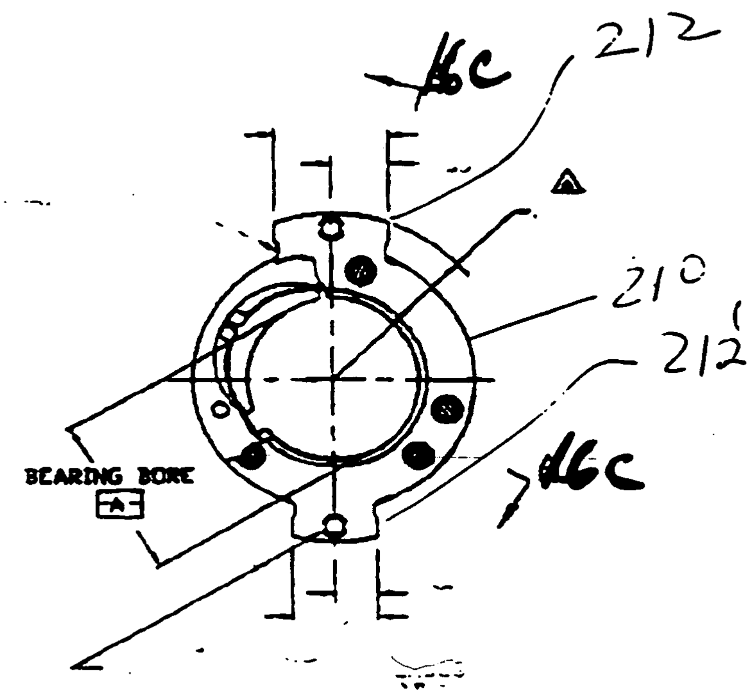


FIG. 16A

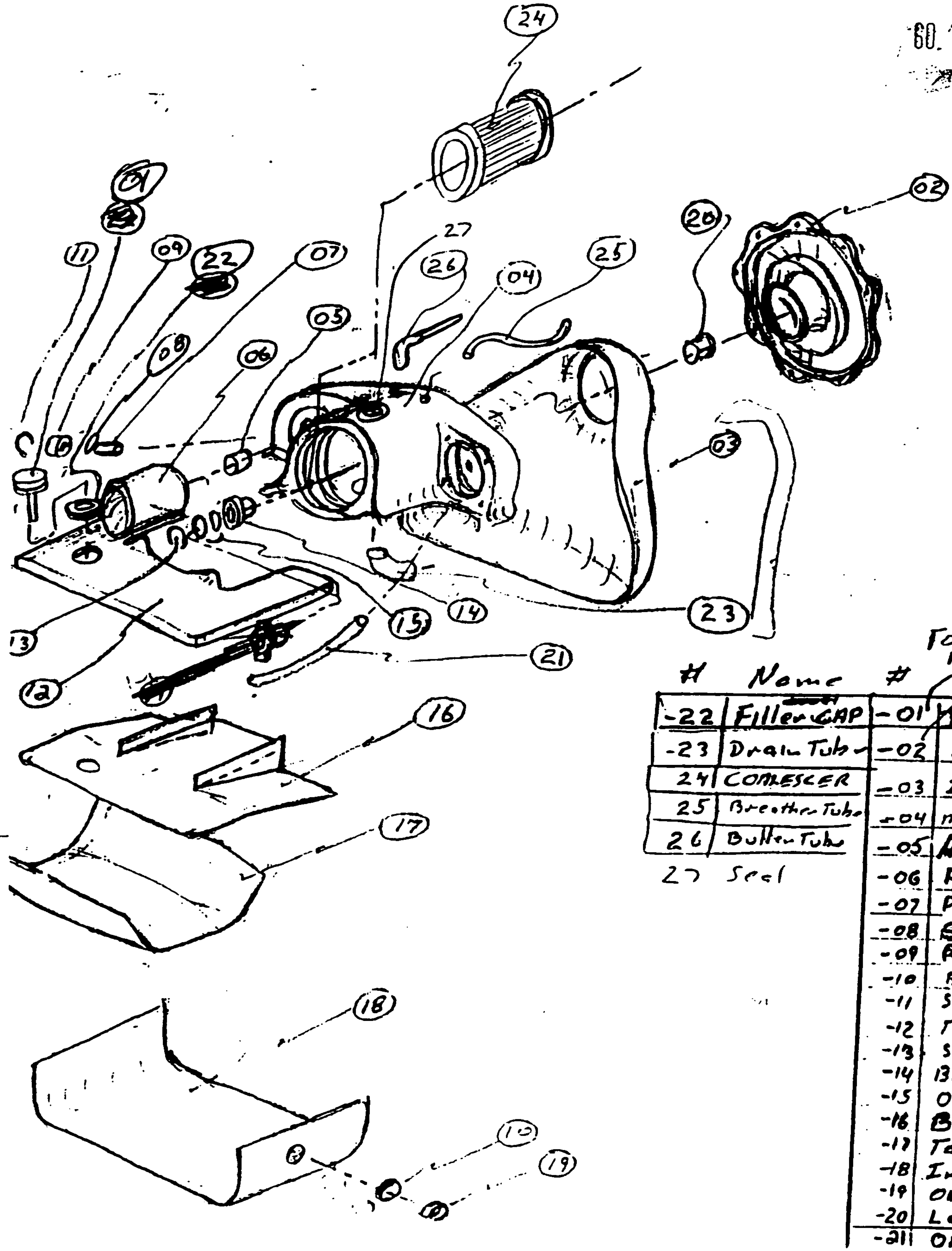
60 032090



END VIEW
(HIDDEN LINES REMOVED)

FIG. 16B

60. 032090



FORD Fill Cap

#	Name	#	Name
-22	Filler CAP	-01	[redacted]
-23	Drain Tube	-02	Bell mouth
24	COALESCER	-03	Intake Shovel
25	Breather Tube	-04	Main Case
26	Butter Tube	-05	Adaptor
27	Seal	-06	Filter, Oil
		-07	Pressure Valve
		-08	Seal
		-09	Retainer
		-10	Boss
		-11	Snap Ring
		-12	Tank Cover
		-13	Snap Ring
		-14	BRG HSG
		-15	O-Rings
		-16	Baffle Plate
		-17	Tank, Oil
		-18	Intake form
		-19	Oil Plug
		-20	Lab. Seal
		-21	Oil Tub

FIG. 17

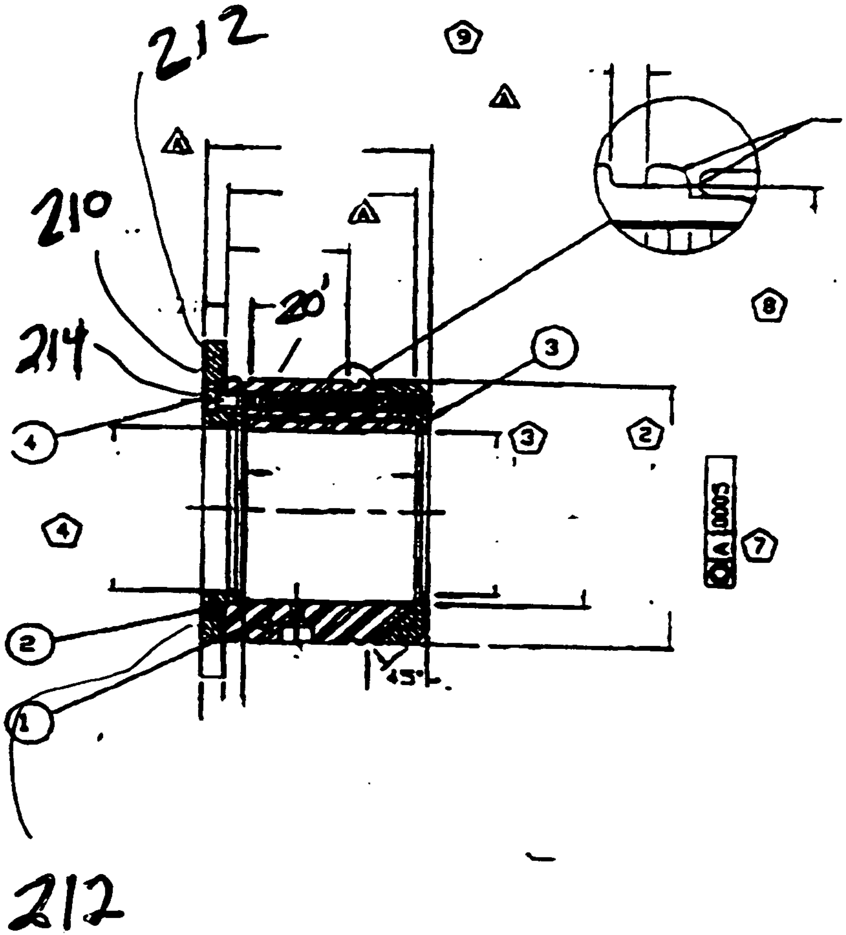


FIG 16C

SECTION A-A

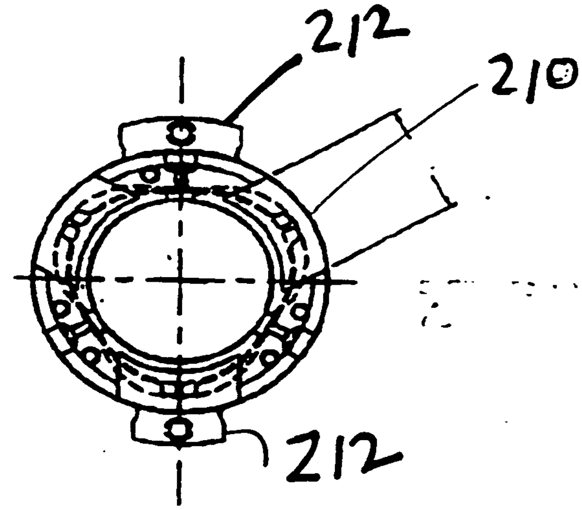


FIG. 16D

60/032090

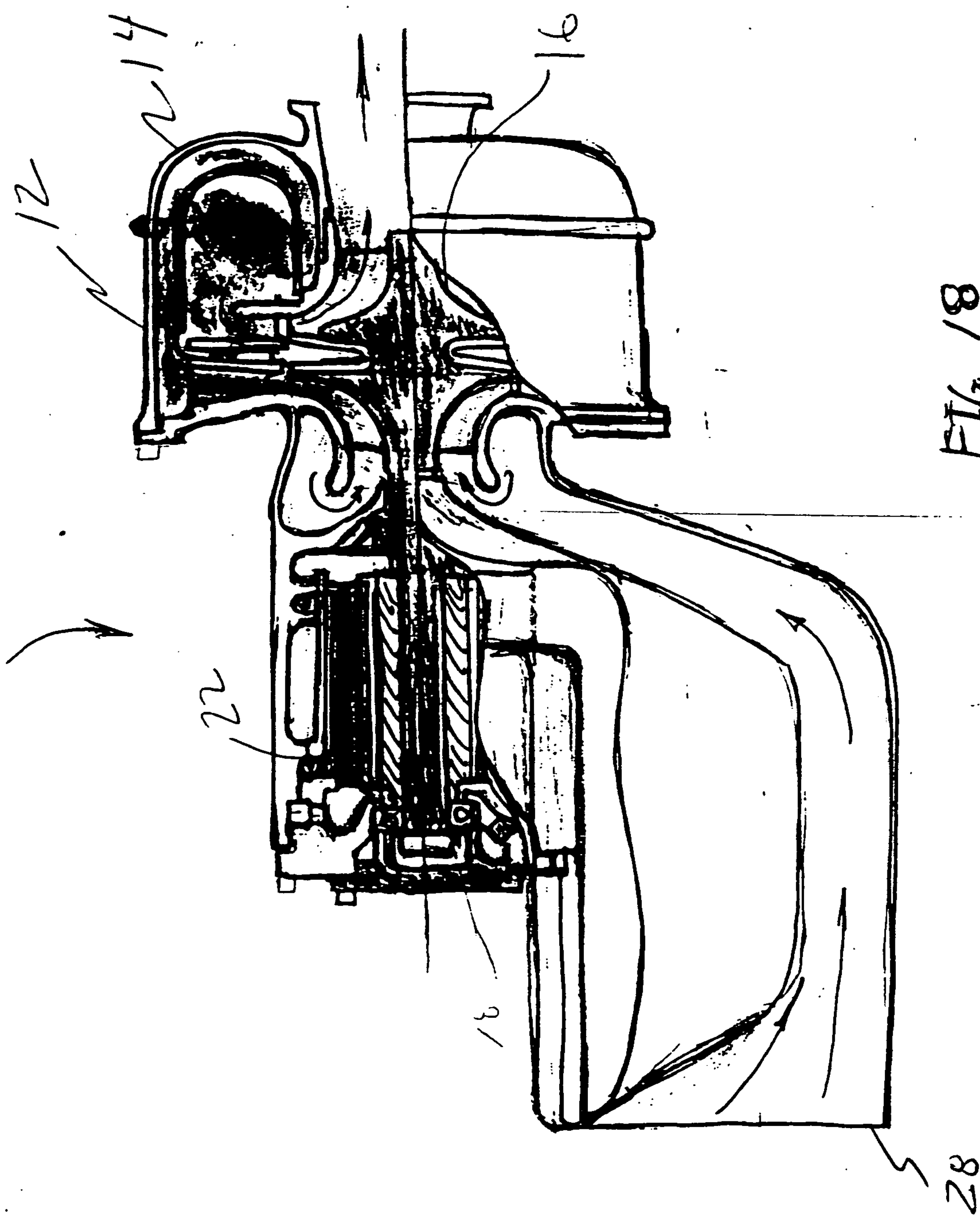


FIG 18

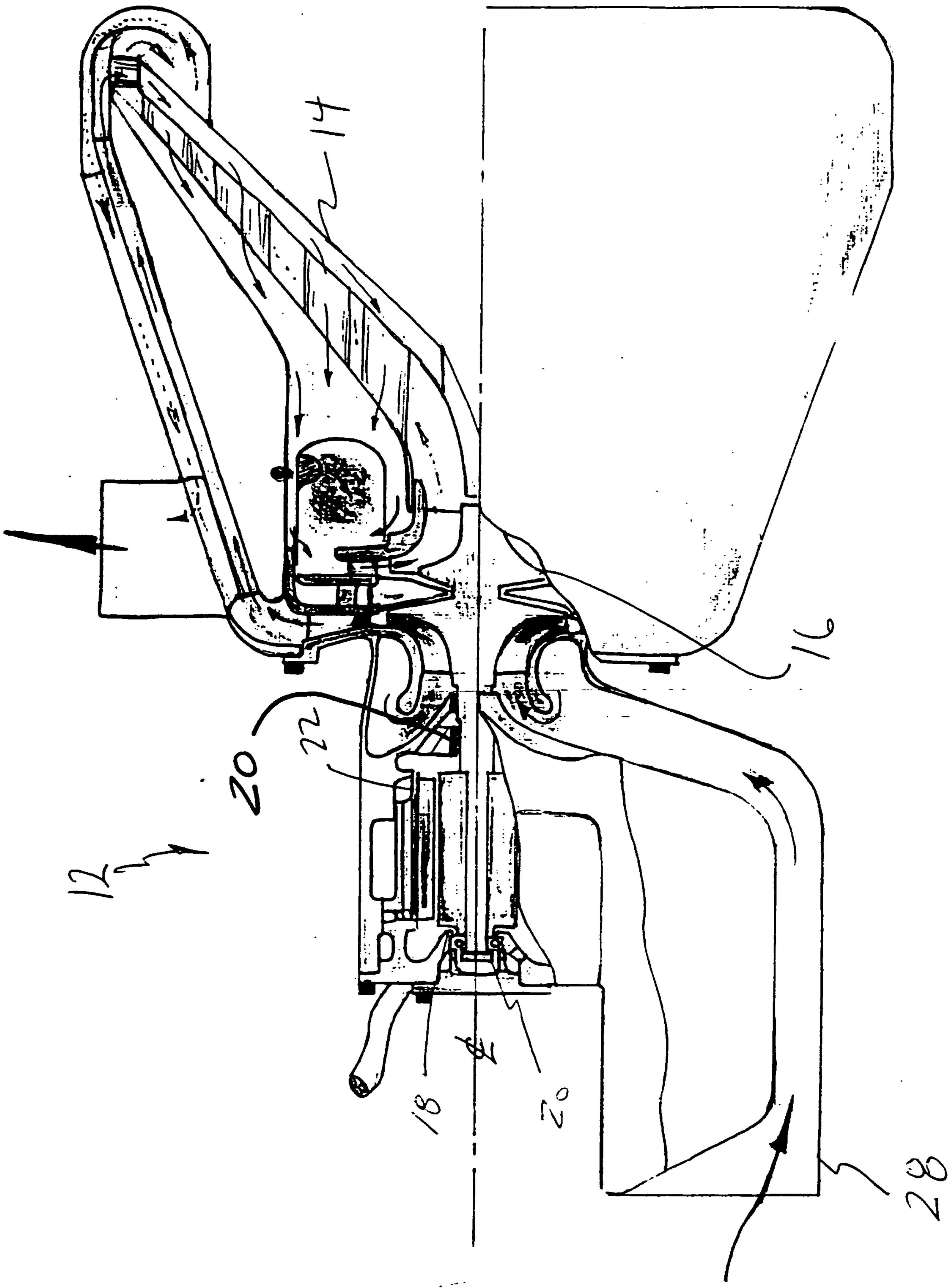
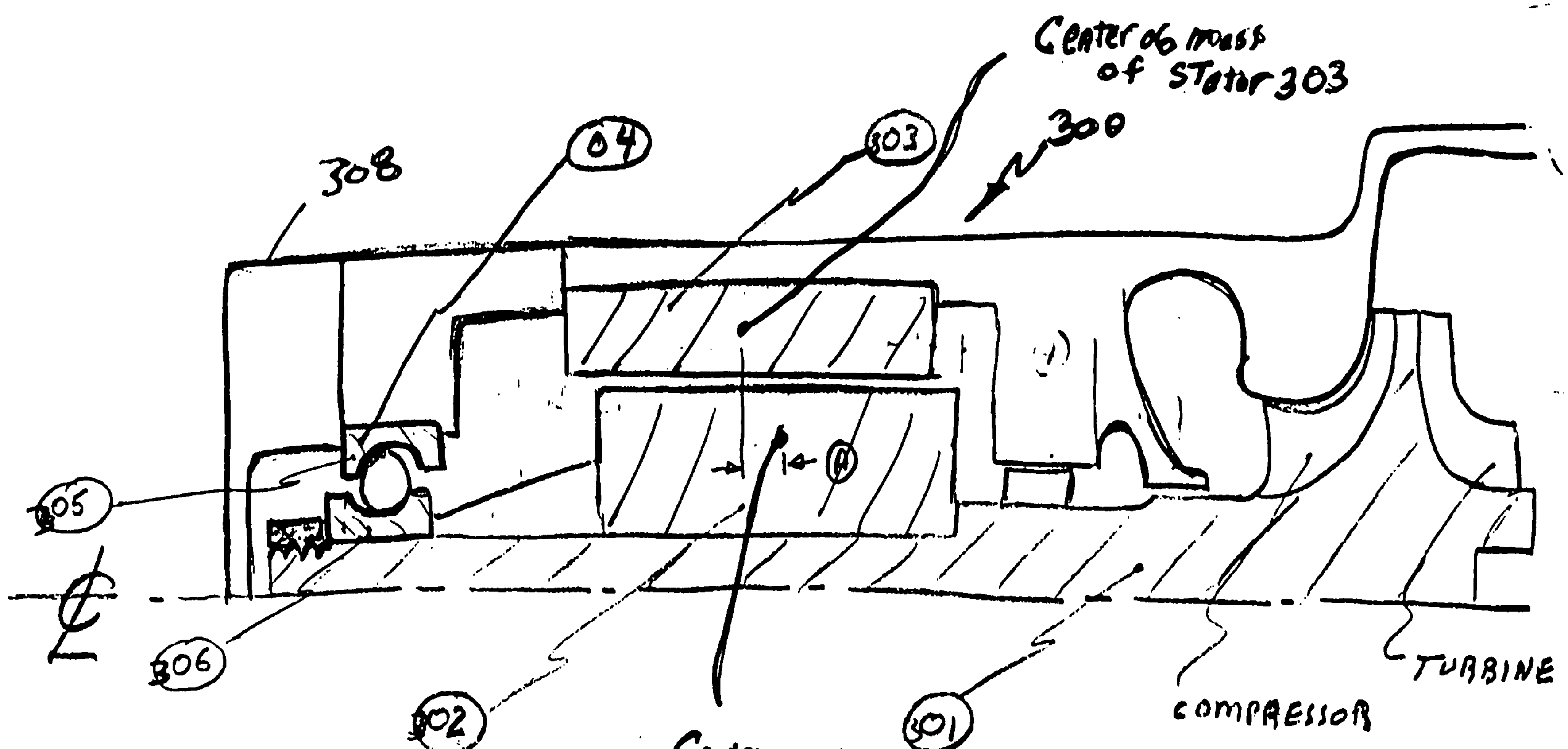


FIG. 19



301 Engine rotor

302 Permanent magnet alternator Rotor

303 Alternator Stator

304 Ball BEARING

305 Outer Race

306 Inner Race

Magnetic Preloaded Ball BRG.

FIG. 2D

60/032090

PATENT APPLICATION SERIAL NO. 60/032090

U.S. DEPARTMENT OF COMMERCE
PATENT AND TRADEMARK OFFICE
FEE RECORD SHEET

260 HP 12/20/96 60032090
1 114 150.00 CK 815960812

PTO-1556
(5/87)

PROVISIONAL APPLICATION COVER SHEET

This is a request for filing a PROVISIONAL APPLICATION under 37 CFR 1.53(b)(2). 60/032090

1147 PTO Attorney's Docket No. 815-960812	Type a plus sign (+) inside this box → +

12/03/96

INVENTOR(s)/APPLICANT(s)	
LAST NAME, FIRST NAME, MIDDLE INITIAL	RESIDENCE (CITY AND EITHER STATE OR FOREIGN COUNTRY)
Teets, Jon W. 100 Teets, Joseph M. 200	~~Scottsdale,~~ Arizona AZ ~~Hobe Sound,~~ Florida AZ

TITLE OF THE INVENTION (280 characters max)

"An Electricity Generating System Having An Annular Combustor"

CORRESPONDENCE ADDRESS

Webb Ziesenheim Bruening Logsdon Orkin & Hanson, P.C.
700 Koppers Building, 436 Seventh Avenue
Pittsburgh, Pennsylvania 15219-1818, United States of America
Telephone No.: (412) 471-8815, Facsimile No.: (412) 471-4094

ENCLOSED APPLICATION PARTS *(check all that apply)*

43 Number of Pages in Specification ____ Small Entity Statement

19 Number of Sheets of Drawing(s) ____ Other (specify)____

METHOD OF PAYMENT *(check one)*

_X_ A check or money order is enclosed to cover the Provisional filing fees ($150.00 Large Entity; $75.00 Small Entity).

_X_ The Commissioner is hereby authorized to charge any additional fees required for filing this application to Deposit Account No. 23-0650. Please refund any overpayment in the form of a check. An original and two copies of this sheet are enclosed.

PROVISIONAL FILING FEE AMOUNT(S) $ 150.00

The invention was made by an agency of the United States Government or under a contract with an agency of the United States Government.

_X_ No.

____ Yes, the name of the U.S. Government agency and the Government contract No. are: ____

Respectfully submitted,

SIGNATURE Paul M. Reznick

TYPED NAME Paul M. Reznick

Date December 3, 1996

Registration No. 33,059

☐ Additional inventors are being named on separately numbered sheets attached hereto.

PROVISIONAL APPLICATION FILING ONLY

"EXPRESS MAIL" mailing label number: EM595290176US

Date of Deposit December 3, 1996

I hereby certify that this paper or fee is being deposited with the United States Postal Service "Express Mail Post Office to Addressee" service under 37 C.F.R. 1.10 on the date indicated above and is addressed to the: Assistant Commissioner of Patents, Washington, D.C. 20231

Nora Ann Pastrick

(Type or print name of person mailing paper)

Nora Ann Pastrick

(Signature of person mailing paper)

19 150-114 A/Prov.

11477 U.S. PTO
12/03/96

60/032090

AN ELECTRICITY GENERATING SYSTEM HAVING AN ANNULAR COMBUSTOR

BACKGROUND OF THE INVENTION

1) Field of the Invention

This invention relates generally to a system for generating electricity, and more particularly, to a compact system which includes an annular combustor and a turbine for generating electricity.

2) Description of the Prior Art

Compact electricity generating systems using annular combustors and turbines are known. Generally, these systems are used to generate between twenty-five to fifty kilowatts of electric power. Such systems are manufactured by Capstone Turbine Corporation, Marbaix, Bowman Power Systems, Ltd. and Allied-Signal Corp.

The majority of the above-described electricity generating systems are designed for use by the military in combat conditions, although they can be used in other applications. Hence, these generating systems are built pursuant to military specifications, which results in expensive systems.

While the military demand for compact electricity generating systems has diminished, there has been a recent interest in these systems for non-military applications, primarily as a backup power source for computers. However, the acceptance of these systems has been limited due to their high cost.

Therefore, it is an object of the present invention to provide an inexpensive, compact durable electricity generating system that includes an annular combustor. This combustor burns hydrocarbon fuels, such as diesel, jet, gasoline, natural gas and alcohol type fuels. Typically, the exhaust gases (of other gas turbines) exiting the combustor must be treated to limit the NO_X emissions exiting into the atmosphere.

Therefore, it is another object of the present invention to provide a low NO_X and general overall low emissions combustor.

Furthermore, in many applications, electricity generating systems of this type are operated intermittently and such use of the systems can cause clogged fuel lines, injectors and/or fuel pumps. It is important that these systems operate on demand because they are primarily used as backup systems for a primary power source.

Therefore, it is a further object of the present invention to provide a reliable electricity generating system that can operate intermittently with consistent reliability.

BRIEF DESCRIPTION OF THE DRAWINGS

Fig. 1 is a schematic diagram of a generating system in accordance with the invention;

Fig. 2 is a schematic diagram of a liquid fuel supply system for the generating system shown in Fig. 1;

Fig. 3 is a schematic diagram of an alternate oil system for the generating system shown in Fig. 1;

Fig. 4 is a plan view of a motor, fuel pump and oil pump arrangement used in the generating system shown in Fig. 1;

Fig. 5 is an end view of a portion of the fuel pump shown in Fig. 4;

Fig. 6 is a side elevation of the fuel pump shown in Fig. 5;

Fig. 7 is a top plan view of a portion of the fuel pump shown in Figs. 5 and 6;

Fig. 8A is a partial section of a metering valve in accordance with the invention;

Fig. 8B is a partial section of the metering valve shown in Fig. 8A;

Fig. 9 is a partial section of another embodiment of a metering valve in accordance with the invention;

Fig. 10 is a section of a portion of the combustor in the generating system shown in Fig. 1;

Fig. 11 is a partial section on line 11-11 of Fig. 10;

Fig. 12 is a top perspective view, partially in section, of a portion of another embodiment of a combustor similar to the combustor shown in Fig. 10;

Fig. 13 is a perspective view of an alternate design of a primary/secondary premixing chamber of the combustor shown in Fig. 10;

Fig. 14 is a graph of flame temperature verses fuel and air mixtures;

Fig. 15 is a partial longitudinal section of a portion of the turbine in accordance with the invention;

Fig. 16A shows an exploded view of a bearing retention system used in the turbine of the invention;

Fig. 16B is a front plan view of a portion of the bearing retention ring and the bearing shown in Fig. 16A;

Fig. 16C is a section taken along lines 16C-16C of Fig. 16B;

Fig. 16D is another front plan view of the bearing retention ring and the bearing shown in Fig. 16B;

Fig. 17 is an exploded perspective view of a portion of the turbine containing the bearing retention system shown in Fig. 16A;

Fig. 18 is a side elevation, partially in section, of the power plant schematically shown in Fig. 1;

Fig. 19 is a side elevation, partially in section, of another embodiment of a power plant shown in Fig. 18; and

Fig. 20 is a side elevation, partially in section, of a portion of a magnetic pre-load ball bearing system made in accordance with the present invention.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

For purposes of description herein, the terms "upper", "lower", "right", "left", "rear", "front", "vertical", "horizontal" and derivatives thereof shall relate to the invention as oriented in the drawing figures. However, it is to be understood that the invention may assume various alternative orientation and step sequences, except where expressly specified to the contrary. It is

also to be understood that the specific devices and processes illustrated in the attached drawings, and described in the following specification are simply exemplary embodiments of the inventive concepts defined in the appended claims. Hence, specific dimensions and other physical characteristics relating to the embodiments disclosed herein are not to be considered as limiting, unless the claims expressly state otherwise.

Fig. 1 of the drawings shows a schematic diagram of an electricity generating system 10 in accordance with the invention. The system 10 includes a power plant 12 having an annular combustor 14 with a combustion chamber through which gas products of combustion pass prior to exiting through an exit port 26. Two specific power plant embodiments are shown in Figs. 18 and 19 of the drawings. The embodiment shown in Fig. 19 of the drawings incorporates a heat exchanger to recoup some exhaust gas heat and improves the overall thermal efficiency of the system. The embodiment shown in Fig. 18 of the drawings does not include a heat exchanger. The annular combustor 14 is fluidly coupled to a turbine rotor 16 which includes a rotor 18 rotatably supported on the opposed ends by bearings 20. A stator 22 is positioned coaxially with the rotor 18 and a heat exchanger 24 is fluidly coupled to the turbine rotor 16. An air inlet port 28 is provided.

Liquid fuel, such as heating oil, is contained in a fuel tank 30 which is connected to the annular combustor 14 by a conduit 32. The conduit 32 is connected to a fuel filter 34, a fuel pump 36 and a fuel metering valve 38. The conduit 32 supplies a plurality of fuel injectors 40 provided in the annular combustor 14. Fig. 2 of the drawings depicts a purge valve 39 and is connected to the conduit 32 between the fuel injectors 40 and the fuel metering valve 38. A conduit 41 connects the fuel purge valve 39 to the fuel tank 30 to discharge fuel to the fuel

reservoir during normal engine shutdown allowing fuel in the injectors and fuel manifold to be purged out and hence, preventing fuel coking/clogging tendencies.

Referring to Figs. 1 and 3 of the drawings, lubricating oil is supplied to the bearings 20 from a sump 42 which is connected to the bearings 20 by a conduit 44. (Fig. 3 of the drawings shows an alternate arrangement from Fig. 1 of the drawings and shows some external engine components coacting with the lubricating oil system, which are not shown in Fig. 1 of the drawings. The arrangement shown in Fig. 3 of the drawings can be incorporated with the generating system shown in Fig. 1 of the drawings.) The conduit 44 is connected to an oil filter 46, an air/oil heat exchanger 48 and a pump 50. Lubricating oil flowing through the bearings 20 returns to the sump 42, along with oil out of the alternator stator heat exchanger 24.

Referring again to Fig. 1 of the drawings, both fuel pump 36 and lubricating oil pump 50 are positive displacement pumps which are mechanically driven by a twenty-four volt electric motor 52. Transducers 54, 56, 58 and 60 are provided for measuring lubricating oil temperature, lubricating oil pressure, fuel pressure and the compressor exit gas pressure, respectively. Transducers 54, 56, 58 and 60 are electrically coupled to a microprocessor controlled engine controller 62. A thermocouple 64 is positioned in the exit port 26 downstream of the turbine for measuring the temperature of the turbine exhaust gases. Thermocouple 64 is electrically coupled to the engine controller 62.

The engine controller 62 is electrically connected to an inverter arrangement 66 that includes an output inverter 68 and a start inverter 70. This arrangement is disclosed in the provisional application entitled "Electrical Control System With Integral Electric Starter/Alternator Coplanar With Turbomachinery Main Rotor" having inventors Suresh Gupta, Douglas R. Burnham, Jon W.

Teets, Joseph M. Teets and Brij Bhargava, filed concurrently herewith. The start inverter 70 is electrically connected to a twenty-four volt DC battery 72 as well as to the engine controller 62 by an input line 74. An output line 76 electrically connects the engine controller 62 to the output inverter 68. The output inverter 68 is adapted to provide electricity by a line 79 to a customer electric supply 83 or to power an electrical component, such as a computer.

Fig. 4 of the drawings shows the electric motor 52 mechanically coupled to the fuel pump 36 and to the lubricating oil pump 50. Preferably, electric motor 52 is a brushless electric motor. The pumps 36 and 50 are operatively connected to the electric motor 52 by rotatable drive shafts 78 and 80, respectively. Energizing the electric motor 52 causes the drive shafts 78 and 80 to rotate about their longitudinal axes 81.

The pumps 36 and 50 are positive displacement pumps and preferably are gerotor type. Referring to Figs. 5-7 of the drawings, each fuel pump 36 includes an inner rotor 82, an outer rotor 84 and a casing 86. An arcuate inlet port 88 and an arcuate outlet port 90 are formed in the casing 86. Drive shaft 78 is mechanically coupled to the inner rotor 82 so that rotation of the shaft about the longitudinal axis 81 causes the inner rotor 82 to rotate relative to the outer rotor 84. The outer rotor 84 defines a plurality (N) of pumping chambers 92 and a plurality (N-1) of radially extending gear teeth 94 which are formed on the inner rotor 82 and are received in the pumping chambers 92 in a manner well known in the art. Specifically, as the inner rotor 82 rotates, liquid (lubricating oil) is pumped from the inlet tube 95 to inlet port 88 through the pumping chambers 92, the outlet port 90 and the outlet tube 96. The lubricating oil pump 50 operates in the same manner as fuel pump 36, with the exception that it is driven by the drive shaft 80 and is not discussed in further detail.

An advantage of the present oil pump/fuel pump motor arrangement is that if the lubricating oil pump 50 fails (which typically means that the inner rotor 82 becomes jammed and cannot rotate about the longitudinal axis 81), the electric motor 52 will stall, thereby preventing the drive shafts 78 and 80 from rotating. Also, if the electric motor or fuel pump fails, there will be a safe shutdown. This causes the system to "shut down" because no fuel will be supplied to the annular combustor 14 by the fuel pump 36 which is driven by the drive shafts 78 and 80. Hence, damage to the system components is prevented due to an inadequate supply of lubricating oil to the rotating system parts. The lubricating oil pump 50 and/or electric motor 52 must be repaired before fuel can be supplied to the annular combustor 14.

Referring to Figs. 1, 8A, 8B and 9 of the drawings, fuel is pumped by the fuel pump and flow varied to the engine by the fuel metering valve 38. Preferably, the fuel metering valve 38 is a spring-loaded closed, proportional solenoid valve. The position of the solenoid valve varies as a function of the current passing through the solenoid which varies the rate of fuel flow through the valve 38.

Figs. 8A (open position) and 8B (close position) of the drawings show one embodiment of the fuel metering valve 38 wherein the valve is designated V. The valve V includes a proportional solenoid S and a valve body B. A longitudinally movable cylindrical plunger P includes a variable diameter tip T. An orifice plate F having a central orifice O is provided in the valve body B. (Alternatively, only the plunger could be used to coact with the orifice O.) The plate F divides the valve body into an inlet chamber and an outlet chamber. A fuel inlet line FI is connected to the inlet chamber and a fuel outlet line FO is connected to the outlet chamber. Activation of the solenoid S causes cylindrical plunger P and tip T to move in the longitudinal direction. The tip T coacts with

the orifice O in the orifice plate F to vary the size of the orifice O, shown in Fig. 8A of the drawings. Fig. 8B of the drawings shows the tip T closing the orifice O to prevent the flow of fuel between the inlet chamber and the outlet chamber. Hence, the position of the tip T relative to the orifice plate F controls the flow of fuel to the annular combustor 14.

Fig. 9 of the drawings shows another embodiment of the fuel metering valve 38 wherein the valve is designated V′. The valve V′ includes a proportional solenoid S′ and a valve body B′. A longitudinally movable cylindrical plunger P′ is provided in the valve body B′. Fuel enters from fuel inlet line FI to a cylindrical chamber on cylindrical plunger P′ that is a continuous ring R_1 around the cylindrical plunger P′. Flow of fuel goes from ring R_1 through a connecting shaft hole H_1 connected to shaft hole H_2 via a hole passageway H_3 then out fuel outlet line FO via ring R_2.

The closed position is when the cylindrical plunger P′ is fully positioned to the left, as shown in Fig. 9. This closes off ring R_2 from the fuel outlet line FO. Metering of fuel occurs by positioning of annular ring R_2 to fuel outlet line FO. Vent lines are also connected to the cavity at the end of the plunger travel areas.

In the operation of the metering valve shown in Fig. 9 of the drawings, the proportional solenoid S′ is activated to move the plunger P′ within the cavity of the valve body B′. The plunger P′ (positioning ring R_2) is then positioned to either block fuel flow from the fuel inlet line FI to the fuel outlet line FO or permit fuel to flow therethrough. The fuel flow rate depends on the longitudinal position ring R_2 relative to the fuel outlet line FO, provided the fuel pump pressure remains constant. Fuel pump pressure to the metering valve is maintained via a pressure relieve valve. The rings R_1 and R_2 are defined on manifold M.

Referring again to Figs. 1 and 2 of the drawings, the fuel purge valve 39 is a normally closed solenoid valve, such as a twenty-four volt DC two-way N.C. solenoid valve. In operation, the purge valve 39 is only in the open position for a fixed period of time when the fuel to the engine (via metering valve) is shut off. Electric motor 52 is still on, until the rotor speed reaches zero RPM at which time the electric motor gets turned off. This allows any residual fuel in the fuel injectors 40 or its related manifold to be blown out by the combustor pressure into the fuel tank 30. This purging operation minimizes/prevents fuel from coking, clogging or plugging the fuel injectors, which can cause problems in fuel distribution.

Fig. 10 of the drawings shows a partial cross section of a portion of the annular combustor 14. The annular combustor 14 is connected to a compressor/turbine arrangement 100. The compressor/turbine arrangement 100 includes compressor blades 102 and turbine blades 104 positioned around a rotary drive shaft 106. Cantilevered from an outboard bearing, the engine rotor or drive shaft 106 is adapted to rotate about a longitudinal Z axis and is supported by bearings 20 which are schematically shown in Fig. 1 of the drawings.

An annular outer housing wall 108 is provided and defines an air intake passage 110 positioned adjacent compressor blades 102. An outer combustor liner wall 112 and a forward housing wall 114 define an annular combustion chamber 116. The forward housing wall 114 and a forward portion of the outer housing wall 108 define a compressor/diffuser air path 118 which begins adjacent to a diffuser exit. An annular cooling area 119 is defined by a distal end of the forward housing wall 120 and a forward end of the outer combustor liner wall 112. An annular air dilution duct 122 is defined at a terminal end of the outer combustor liner wall 112. A corrugated wiggle strip 124 is provided in the air dilution duct 122.

An upwardly extending curved turbine nozzle wall 126 is spaced from air dilution nozzle 122. The turbine nozzle wall 126 and the forward housing wall 114 define an annular turbine nozzle 128 which is in fluid communication with the turbine blades 104. An air flow path 129 is defined between the outer housing wall 108 and the outer combustor liner wall 112.

A plurality of premix chambers 130 are circumferentially spaced about the outer combustor liner wall 112 adjacent to a rearward wall of the combustion chamber 116. A plurality of circumferentially dispensed radially positioned fuel injectors 132 extend through the outer housing wall 108 and into the flow path or cavity 129 such as to position fuel delivery to the primary premix chamber or inlet zone 138 of Fig. 11.

Referring to Fig. 11 of the drawings, the fuel injectors 132 pass through the outer housing wall 108 and terminate within the air flow path 129. A plurality of primary premix conduits 134 extend circumferentially about the outer combustor liner wall 112 adjacent to the back wall 136 of the annular combustion chamber 116. Inlet zone or ends 138 of the primary premix conduits 134 are positioned in close proximity to the terminating ends of the fuel injector 132 and are angled so that they face in the flow direction of arrows 140. A swirler 142 is provided in each of the primary premix conduits 134 (to assist in fuel vaporization). Alternatively, swirlers 142 can be eliminated. Primary premix conduits 134 are arranged with respect to the outlet ends of the fuel injectors 132 to direct a rich (non-combustible mixture) air fuel mixture in a circumferential axial direction toward the forward housing wall 114 of the annular combustion chamber 116. An igniter 144 is provided in the outer combustor liner wall 112 to ignite the fuel/air (F/A) mixture creating a self sustaining flame.

The operation of the combustor is described hereinafter with reference to Figs. 10-11 of the drawings.

The engine rotor 106 is rotated causing compressor blades 102 to rotate about the Z axis. Air is drawn into the intake 110, the air path 118 and the flow path 129 in the direction of arrows 140. The directed air exits into the combustion chamber 116 through the cooling duct 119 and the air dilution nozzle 122. Air also enters the inlet ends 138 of the primary premix conduits 134. Pressurized fuel exits the ends of the fuel injectors or nozzles 132 and is carried by the (because of generated differential pressure across the combustor liner) air into the inlet ends 138 of the primary premix conduits 134 forming a rich fuel/air mixture. This fuel/air mixture passes through the optional swirlers (to enhance hot wall fuel vaporization) causing it to swirl. Also, longer conduits 134 can be provided for a higher residence time of a rich fuel/air mixture (outside of auto ignition) will suffice in good vaporization and homogenous fuel/air mixing. Fig. 12 of the drawings shows another embodiment having swirlers 142, as the mixture moves along the primary premix conduits 134. This rich fuel/air mixture then enters premix chambers 130 and then the secondary premix chambers where further air mixes to yield a lean fuel/air mixture for combustion and exits into the annular combustion chamber 116 in a predominantly circumferential direction into the flame front. The igniter 144 ignites the mixture which burns to produce energy for power. Downstream and prior to the air dilution nozzle 122 enters the flame to reduce the temperature. The exiting gases then pass the generated flame front after the dilution air is mixed and goes into and through the turbine nozzle to generate a velocity for related turbine wheel power extraction through the turbine blades 104, which drives the compressor wheel/blades 102 and the alternator shown in Figs. 18 and 19 of the drawings.

Fig. 13 of the drawings shows an alternate arrangement of the previously described premix chambers 130. Specifically, each primary premix conduit 134 is fed into a twisted lobed fuel secondary premix chamber 150 to

enhance the secondary premixing prior to combustion. Each of the lobes 152 has a twisted shape to cause the fuel/air mixture to swirl. Secondary air conduits 154 are provided and connected to the secondary premix chamber 150. The entry ends of the secondary air conduits 154 are positioned within the air flow path 129. A mixing block 156 is located at the junction of each primary premix conduit 134, the secondary air conduit 154 and the secondary premix chamber 150 to mix the effluents from conduits 134 and 154. The blocks 156 offer a large mass detail and are attached to the combustor liner and correspondingly yields reduced liner heating and typical distortion tendencies.

In operation, the present invention results in a low NO_x formation and overall emission species are reduced.

Low NO_x ($NO + NO_2$) under 10 ppm is desirable in combustors and can be achieved through a low oxidizing environment (fuel/air rich primary premix, long residence time) with low flame temperatures after the secondary fuel/air lean fuel/air, low residence prep. A long residence time in the primary premix to liberate the nitrogen atom with minimal oxygen available is necessary (rich fuel/air mixture, long residence primary premix). Too low of an oxidizing primary flame zone temperature will cause excess UHC (unburned hydrocarbons) with CO (carbon monoxide). This is why a non-flame primary premix is good. A low operation temperature range flame is attained through a homogenous, lean prevaporized premixed stagged operation. The low flame temperature can be achieved through a fuel rich or fuel lean condition.

Preferably, a rich fuel/air premix prevaporized primary mixing system without a flame is followed by a secondary mixing system to attain lean fuel/air ratios prior to combustion to yield a low flame temperature less than 2500°F and reduced emissions. The rich fuel/air ratio (non-combusted) passes through a short residence time secondary premix leaning stage prior to combustion thus avoiding the stoichiometric flame state and related high

NO_x. Circumferential mixing and burning combined with a primary rich vaporizing premix followed by a secondary premix chamber, lean fuel/air prior to combustion provides a low emissions combustion.

Liberated hydrogen in the rich fuel/air primary premix stage combined with relatively low changes in pressure or pressure drop (ΔP) across the combustor enhances lean flame stability of which the lean secondary chamber short residence leads to.

Initially, in operation, the engine rotor is driven by the battery power, while simultaneously fuel is supplied to the combustion chamber and the igniter is activated. Air flow exits the compressor diffuser in a progressive tangential direction and moves in the direction of the primary injection mixing tubes wherein an amount of air together with low pressure liquid fuel is injected near the entrance of the mixing tubes and by a change in pressure or kinetic energy enters a swirler zone. A simple cavity swirler accepts fuel in two areas to assist in a homogenous mixing from a single jet or a small spray cone fuel supply. The fuel is caused to flow into the mixing tubes by a change in pressure across the combustor liner. The fuel is centrifugally spun at the inner diameter wall of the primary mixing tubes where it is vaporized once a flame has been initiated. Then, the rich vaporized fuel/air mixture exits into the secondary premix zone where fuel/air is leaned prior to exiting to the igniter in the path of the fuel/air mixture and ignites the mixtures. Once a flame has been initiated outside the tube, the heat creates vaporization of the fuel/air mixture within the tubes.

This rich fuel/air mixture in the primary zone which in turn is leaned in the secondary chamber, varies in concentration and flame temperature depending upon engine operational speed; but is in the range of 2700°F to 1500°F where NO_x is minimized.

The combustion after the secondary premix zone has an elevated lean flame temperature with a low equivalence ratio yielding low emission via low temperature and added oxygen reaction for a chemical reaction change of ($CO + OH = CO_2 + H$) reducing the carbon monoxide emission ideally 0.6 to 0.9 ϕ, the latter of which is compromised for a lower flame temperature to keep a low NO_x value.

The combustion products pass through the combustor circumferentially/tangentially keeping the kinetic energy direction typical to that out of the diffuser. The flame enters the dilution zone where further compressor discharge air mixes with the combustor products to reduce the flame temperature to a designated turbine inlet temperature. The fuel/air ratio is dependent on the power requirement and air flow, the latter of which may be constant. Fuel flow varies depending on the applied load to the turbine rotor.

Fig. 14 of the drawings shows some "text book" operating ranges dependent on fuel/air ratios prior to combustion, wherein the stoichiometric temperature of 3800°F flame temperature would yield excessive NO_x. Preferably, the operating temperature is between 1500°F to 2700°F, where the lower level of 0.4 ϕ would be ideal. Compromising with no variable geometry approximately 0.6 ϕ may be best and will vary dependent upon power requirement.

Another important feature of the present invention is the bearings which support the turbine rotor at speeds in excess of 100,000 rpm. Fig. 17 shows the current number of details which make up the engine with details of Fig. 14 depicting to bearing position/location of Fig. 16A. Figs. 15, 16A-16D and 17 of the drawings show a bearing 20, which is a floating and oil damping bearing, that rotatably and slidably receives the engine rotor 16 of Fig. 18 of the drawings. With specific reference to Fig. 16A-16D of the drawings, the bearing 20 includes a

cylindrical one-piece pad type bearing 20'. Two axial extending screw holes are located on one end surface of the bearing 20'. Bearing 20' is received in a bearing housing 200 located in the turbine engine housing 202. Two spaced apart arcuate lips 204 extend axially from one end of the bearing housing 200. Arcuate-shaped grooves 206 (of which only one is shown) are defined on the inner circumferential surfaces of the lips 204. Spaced lug receiving recesses 208 are defined by the ends of lips 204. An annular retention lugged ring 210 is provided adjacent to the end of the bearing 20' having the screw holes. Two lugs 212, spaced 180° apart, extend radially from the retention ring 210 and two screw receiving holes are located in the retention ring 210 to secure the retention ring to the end of the bearing 20' by screws 214 which pass through the holes in the retention ring into the holes in the end of the bearing. This bearing is then received in the bearing housing 200 with the lugs 212 positioned within the lug receiving recesses 208. A snap ring 216 is inserted in the grooves 206 in the bearing housing 200 to hold the lugs 212 and, in turn, the retention ring 210 between the snap ring 216 and the bearing housing 200.

Preferably, there is a small clearance between the bearing housing 200 and the outer diameter of the bearing 20'. This arrangement provides a full non-interrupted bearing float without the problem of screws coming loose since the snap ring 216 holds the bearing in place. This arrangement also allows for controlled axial and circumferential movement of the bearing 20' while the snap ring 216 restrains the bearing 20' in the axial direction in the bearing housing 200.

A magnetic pre-load system is provided, as shown in Fig. 20 of the drawings. Typical oil lubricated ball bearing systems like to have a light "pre-load" to assure the balls are in contact with respective inner and outer races to prevent relative skidding and inherent material spalling damage during rotor spin up. Gas turbine engines

usually develop a safe bearing thrust load through engine operation pressures at approximately 30% design speed; but until then the balls are subject to some levels of skidding which could lead to "spall" damage. Some small gas turbines have a set of ball bearing spring pre-loaded to each other like machining spindles but the gas turbine may be compromised by inherent poor rotor designs yielding other problems.

The present embodiment includes an integral alternator to engine rotor system 300 which includes a rotor 302 and a stator 303 having an approximate 2% A axially central mass relative displacement as noted creating an inherent axially forward rotor 302 attraction to the stator 303 and thus a beneficial pre-load without alternator electrical output impairment and incorporating only one ball bearing.

Specifically, the alternator rotor 302 includes a permanent magnet which is positioned adjacent an iron stator 303 having centers of masses which are offset by "A". The rotor 302 is attached to the engine rotor 301 (which corresponds to rotor 18 in Fig. 1). A ball bearing 304 is provided on an end of the rotor 301. The ball bearing includes an annular inner race 306 secured to the rotor 301 and an annular outer race 305 coaxially positioned with the inner race 306 and secured to a stator housing 307. Balls 308 are received within a ball receiving recess defined between the inner race 306 and the outer race 305. The magnetic attraction of the stator 303 to the rotor 302 causes a continuous pre-load to be applied to the bearing 304 to prevent spalling.

The electricity generating system 10 operates as follows. First, the generating system 10 is started by drawing energy from the DC battery 72 and a manual fuel valve is opened. This valve is always open and is only closed in emergency situations wherein the fuel must be cut off. The igniter is energized. Power from the DC battery 72 is pulsed to the igniter. The battery power causes the

compressor shaft to rotate so that inlet air flows to the annular combustor 14. The purge valve 39 is maintained in a closed position and is only opened upon shutdown for one minute to purge fuel from fuel injectors 40 to the supply tank 30 by the combustor back pressure.

The electric motor 52 is then energized. This motor drives the lubricating oil pump 50 and gerotor fuel pump 36. The gas turbine engine alternator/motor will not be energized until oil pressure reaches a set minimum. The oil pressure transducer monitors oil pressure to determine emergency shutdown conditions where oil pressure drops below a set level. The fuel pump 36 simultaneously provides a regulated fuel supply pressure.

With the above-noted sequences, the engine stator begins engine rotation causing air to flow into the engine. At 5% design speed, ignition continues and when the engine rotor is at approximately 10% speed, fuel is delivered to the combustor. At 40% minimum design speed the starter is turned off. The engine continues to accelerate to design speed. The igniter 144 ignites the fuel air mixture in the annular combustor. It is important that the ignition of this mixture occurs early to permit a gentle ignition. Initial amounts of fuel flowing to the combustor are established based upon the inlet and exit exhaust gas temperatures which are used to set the proportional solenoid fuel metering valve 38. Following initial ignition when the temperature exceeds 1000°F, the rotor speed accelerates to approximately 90% full speed. The speed of the rotor is dependent on the exhaust gas temperature which is preferably 800°F to 1000°F. The electric motor 52 is shut off if the exhaust temperature exceeds 1500°F for more than four seconds.

Appendix A hereto shows flow charts describing the above-identified operation of the electricity generating system 10. The Appendix is self-explanatory, however, it is important to note that the engine controller continually samples the oil pressure, fuel pressure, oil

temperature, compressor inlet temperature and compressor outlet pressure for efficient operation of the turbine and minimal emissions. Further, the engine controller monitors the system to determine whether there has been a failure of the fuel pump, the oil pump or the electric motor 52 driving these pumps. With reference to Appendix A, "EGT" means exhaust gas temperature; "OST" means oil sump temperature; and "spin down time" means the time the engine rotor takes to "spin down" once the fuel supply is cut off.

60/032090

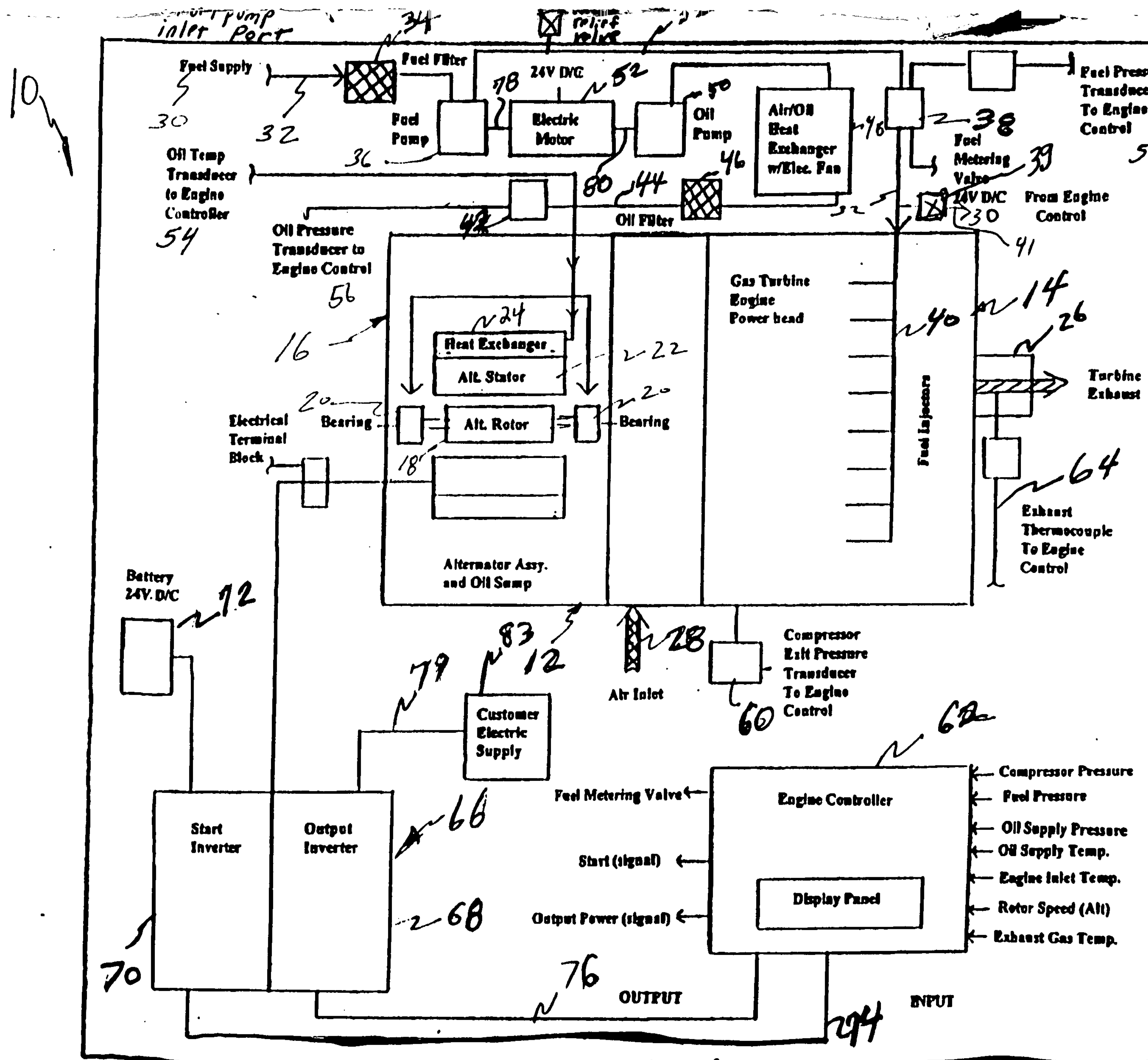


FIG 1

60/032090

LIQUID FUEL SYSTEM

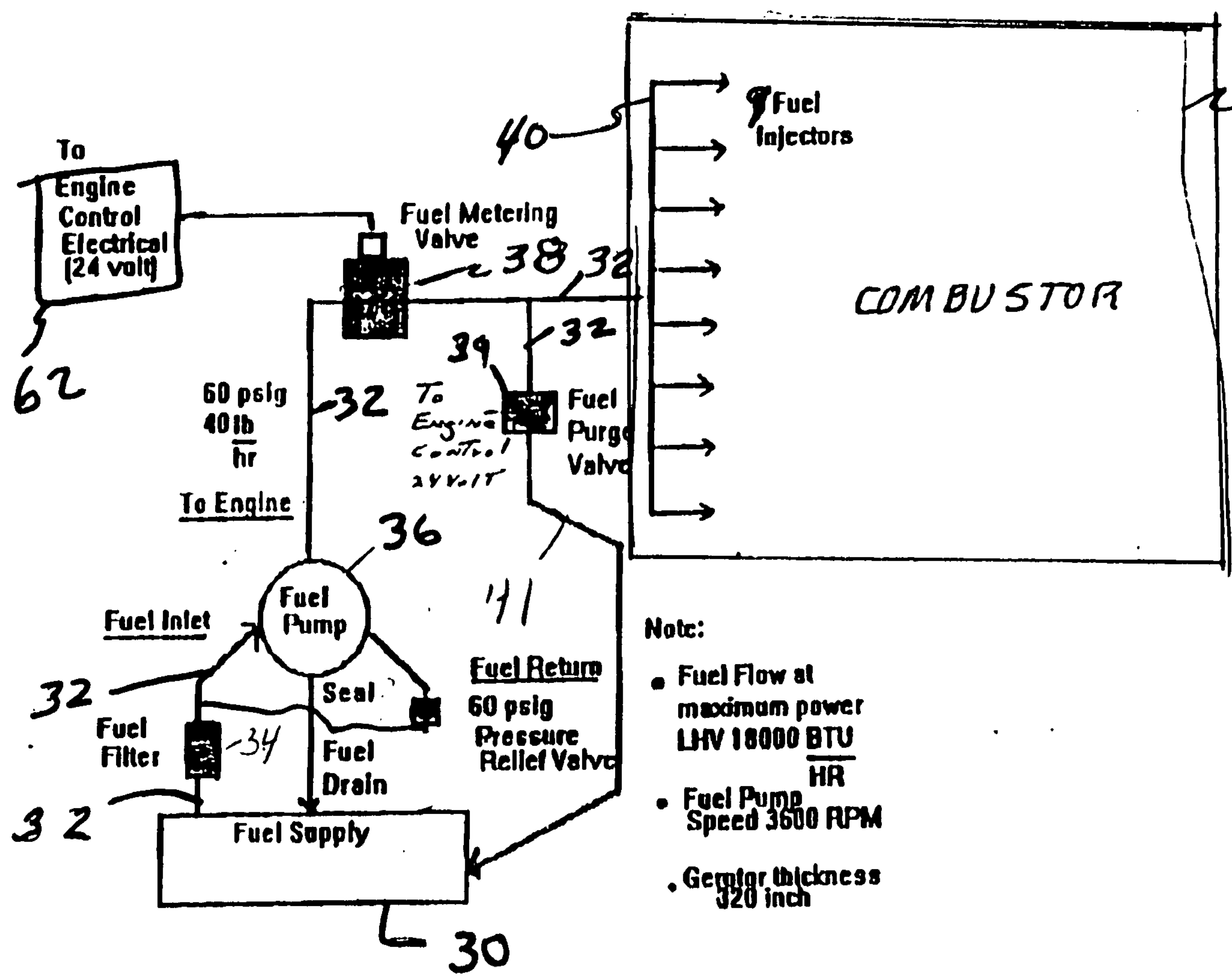


FIG 2

TURBINE ENGINE
OIL SYSTEM

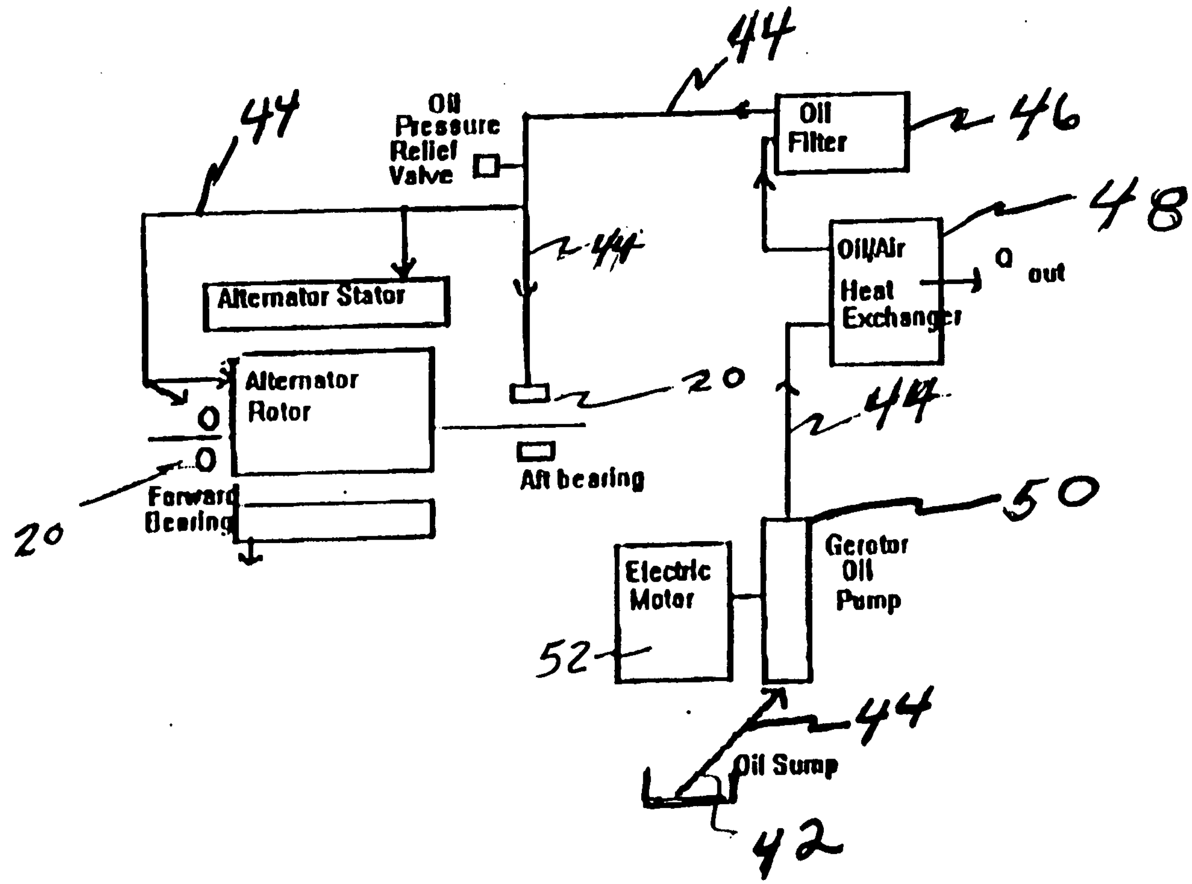


FIG 3

60/032090

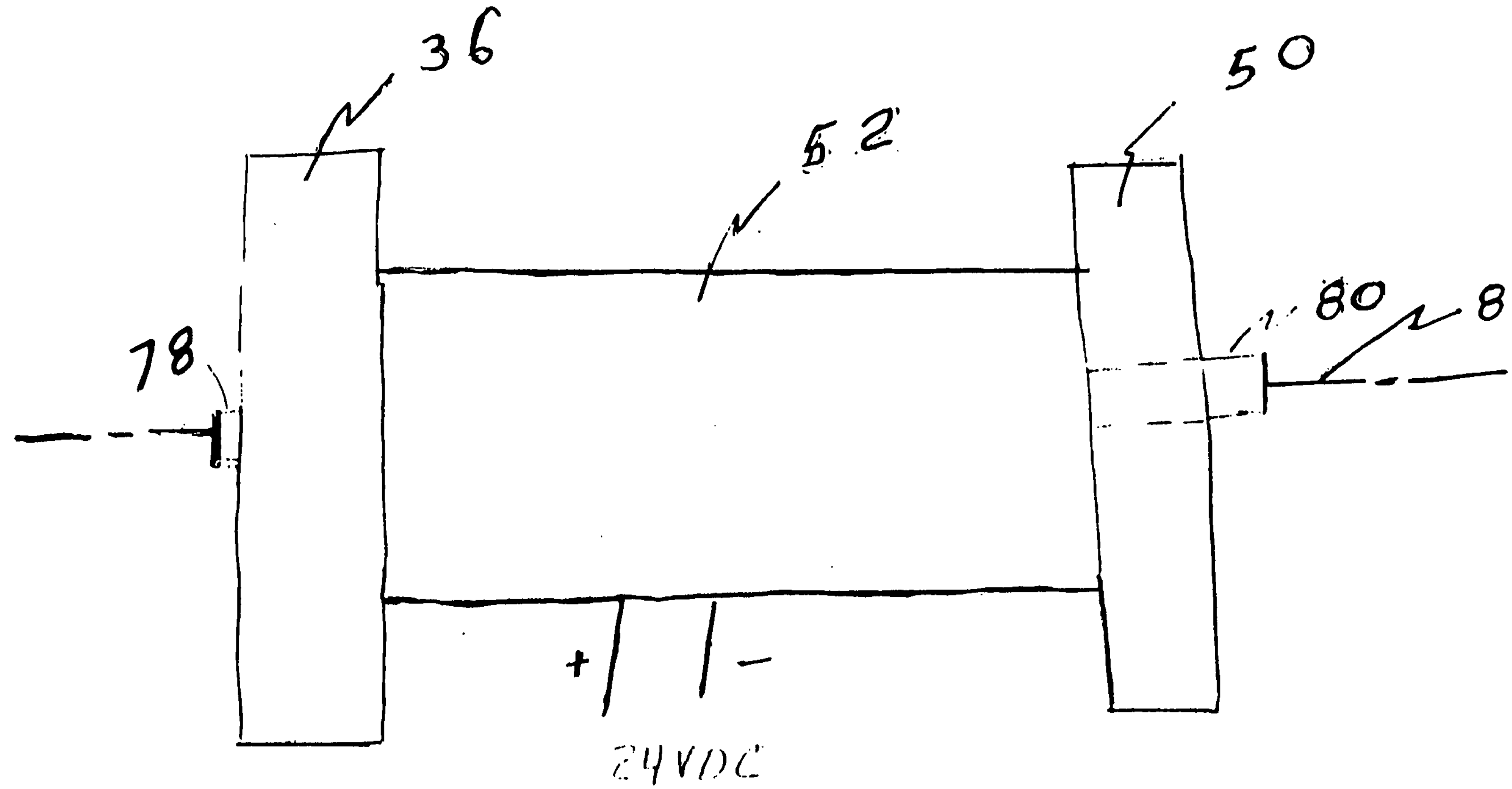


FIG 4

60/032090

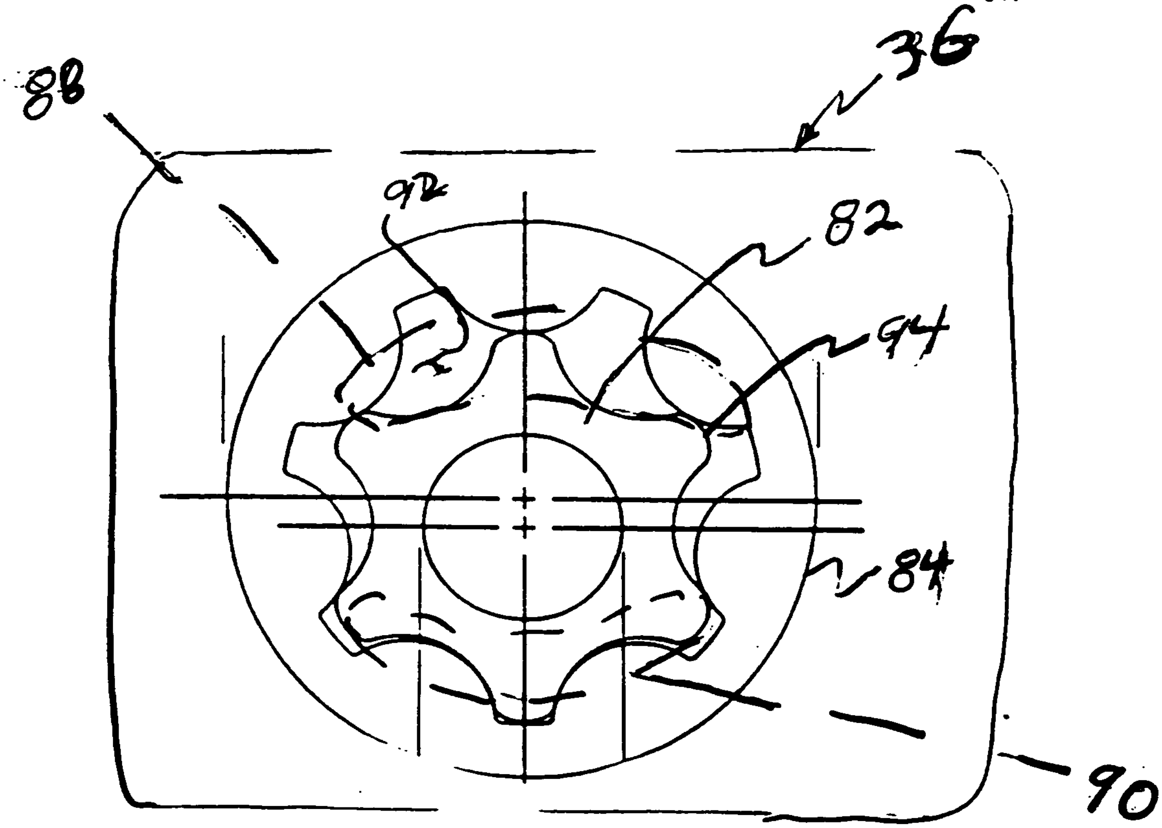


FIG. 5

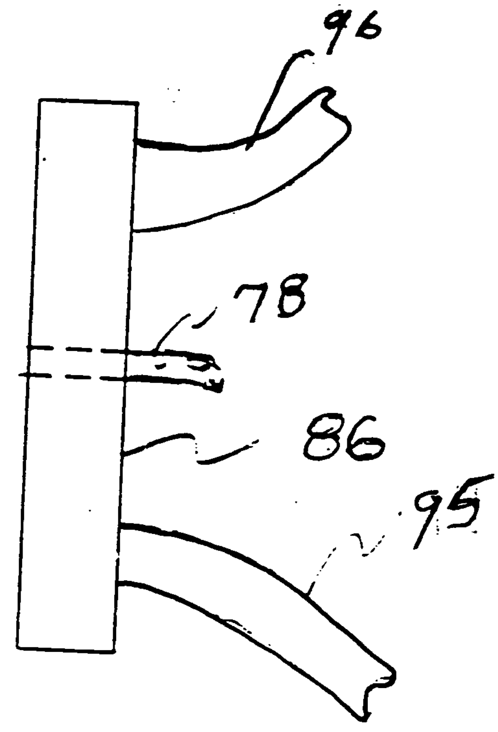


FIG. 6